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THE INSTITUTE OF PETROLEUM

AN Ordinary General Meeting of the Institute of Petroleum was held at 61 New Cavendish Street, London, W.1, on 6 April 1960, the Chair being taken by H. Hyams, Vice-President of the Institute.

The General Secretary read the minutes of the previous meeting, which were confirmed and signed as a correct record.

The Chairman introduced Mr Alban, who then presented the following paper in summary.

THE SYSTEMATIC PREDICTION OF REFINERY UTILITIES *

By F. P. ALBAN †

INTRODUCTION

UTILITIES are defined as the services supplying the necessary mechanical energy, heating, and cooling flows to operate a process unit and maintain it in thermal balance. The main services with which we are concerned are steam, electricity, water, fuel, and compressed air, each of which can be further classified and sub-divided under " non-recoverable " and " recoverable " headings. They may also be generated wholly or in part within a unit or refinery complex or supplied from an external source.

For a simple process study it may not be possible or desirable to investigate a problem from an engineering angle, and a rough appraisal of utility requirements can be made by prorating from current information on similar units or processes, but such studies are primarily a process responsibility, and will not be considered here.

More generally, detailed studies are required which will form the basis for a competitive estimate for a process unit or complete refinery project, where the design development and subsequent cost estimate required to produce the lowest overall refinery cost can represent a considerable financial outlay.

The purpose of this paper is to study a typical organization for the collection and routing of data used for calculating horsepower requirements and for sizing equipment generally, also in the correlation of the resulting information to enable a realistic prediction of utility consumptions to proceed in line with the preliminary design work inherent in the preparation of an estimate of this type.

The many factors which influence the reliability of the predicted quantities are discussed, also the danger of overdesigning, which can result from a too prodigal use of safety factors. To obtain realistic totals, these factors must be rigorously controlled, and this can best be done by a centralized specialist group forming an integral part of the main contracts organization.

The fact that utilities are synonymous with direct operating cost has not always been fully appreciated, and consequently the choice of a process or a contractor may well be decisively influenced by the accuracy of this important phase of estimating.

CLASSIFICATION OF UTILITIES

Utilities fall under three main categories which are kept separate in the preparation of the summaries.

1. Normal utility consumptions which are confined to those consumed directly in the process when fulfilling the quantity and quality process guarantees.
2. Intermittent utility consumptions, which include requirements for standby and infrequently operated equipment, space heating, also regeneration and overhaul requirements.
3. Emergency utilities for dealing with plant upsets and fire fighting.

Table I lists the main utility users likely to be encountered in refinery practice under the three categories, and provides a ready means of checking that everything has been included. Normal consumptions are by far the most important for telling how much it is going to cost to run the plant, and the method of predicting these will be discussed from the basic driver calculations to the final summary.

The second and third categories do not in any way contribute to the day-to-day operating costs, and arbitrary values based on experience are employed in their assessment.

SOME FACTORS INFLUENCING UTILITY PREDICTION

First of all there must be a clear idea of what is meant by normal and design quantities and to what extent these are influenced by the various factors.

The normal or " no-loss " quantities are calculated from the theoretical process material and heat balances shown on the flowsheet without the application of any factor. They form the basis for predicting the normal utility consumptions and the corresponding direct operating cost. The design quantities are used

* MS received 28 March 1960.

† Kellogg International Corporation.

for sizing equipment and piping, and include any process factor added to the " no-loss " figures.

These process factors vary from zero to as much as 40 per cent or more, and are largely dictated by

TABLE I

Check List of Utility Users

Description	Type of operation		
	Normal	Intermittent	Emergency
Steam			
Process (non-recoverable)	×		
Reboilers	×		
Generators (process)	×		
Independently fired boilers	×		
Waste heat boilers	×		
Drivers and (stand-bys)	×	(×)	
Ejectors	×	×	
Special users	×	×	
Oil burners (atomizing)	×	×	
Plant steam		×	
Tracing steam		×	
Space heating		×	
Tank heating		×	
Regenerative processes and overhaul		×	
Emergency and fire fighting			×
Water			
Process (non-recoverable)	×		
Process condensers and coolers	×		
Make-up to closed systems	×	×	
Equipment cooling (bearings, glands, and jackets)	×	(×)	
Surface and jet condensers	×	×	
Compressor inter- and after-cooling	×	(×)	
Plant water		×	
Drinking water		×	
Quench water		×	
Boiler feed water	×		
Condensate	×		
Emergency and fire fighting			×
Power			
Drivers and (stand-bys)	×	(×)	
Lighting	×		
Instruments	×		
Power generation	×		
Emergency power generation			×
Air			
Process	×		
Instrument	×		
Plant		×	
Air motors	×		
Regeneration processes and overhaul		×	
Inert Gas			
Process	×		
Electrical equipment and drivers	×	(×)	
Regenerative processes		×	
Emergency			×
Fuel			
Drivers and (stand-bys)	×	(×)	
Burners	×		

" know how " based on previous experience plus the discretion of the process engineer. They are added to cover situations where the duty of a particular equipment item cannot be calculated with absolute certainty, and may also reflect the design philosophy of a particular refinery or contractor.

In addition to normal and design values which appear on flowsheets, reference may be made to other factors inherent in the design, such as allowance for leakage at compressors and equipment handling light hydrocarbons at high pressure and leakage in refrigeration circuits.

Multi-case operation will occur when provision for alternate types of feedstock, different conversion levels, and product specification is incorporated in the design of a particular unit or if a unit is to operate initially at a stated reduced capacity. Normal utilities are assessed for each case, and will be directly influenced by any intermittent or fractional loading of individual equipment items dictated by the process.

Seasonal variation will generally affect only space and tank heating, but in exceptional cases, where seasonal ambient temperatures vary by 100° F or more, thermal balances will be appreciably altered by large variations in feedstock and cooling water temperature.

Design factors generally do not affect the normal utility values except to the extent that efficiencies are reduced when operating at fractional loads with oversized equipment.

THE BASIC CALCULATION OF UTILITY VALUES

During the early stages of an estimate, and within a few days following process release, it will be necessary for the specialist divisions to arrive at preliminary values of steam and power consumptions for drivers.

The calculations from which these values are derived are based on fundamental equations with which the industry as a whole is familiar. For the purpose of this study, these are confined to the types of pumps, compressors, and drivers which are likely to be main consumers in a refinery, namely,

Centrifugal compressors
Reciprocating compressors
Centrifugal pumps
Electric motors
Steam turbines
Diesel engines
Gas turbines

Simple definitions of the terms used in the equations which follow are given below:

n = polytropic exponent of compression;
k = ratio of specific heats;
P_1 = suction pressure, psia;
P_2 = discharge pressure, psia;
V_1 = inlet volume, cu ft/min;
E = polytropic efficiency;
H = head in feet;

R_o = universal gas constant = $\frac{1545}{M}$;
M = molecular weight;
W = compressor capacity, lb/min;
z = compression ratio = P_2/P_1.
r = compressibility factor;

Reciprocating Compressors

A single-stage compression may be assumed to be isentropic, as during a single stroke of a machine the compression is so rapid by comparison that practically no heat is removed. In multi-stage machines, gas temperatures after cooling are usually 10° above the cooling water temperature. The normal procedure is to assume multi-stage operation with perfect intercooling. The fundamental equation for horsepower is represented by:

(*a*) For single-stage compression—

Theoretical horsepower =

$$\frac{144}{33000} \cdot \frac{n}{n-1} z P_1 V_1 \left[R^{\frac{n-1}{n}} \quad 1 \right]$$

(*b*) For multi-stage compression (x stages)–

Theoretical horsepower =

$$x \cdot \frac{144}{33000} \cdot \frac{n}{n-1} z P_1 V_1 \left[\sqrt[x]{R^{\frac{n-1}{n}}} - 1 \right]$$

n value will be slightly less than k, due to heat loss in the cooling system, radiation, and conduction. It is assumed that $n = 0{\cdot}97k$.

To the compression horsepower figures must be added a percentage for gas losses (valves, pipe friction) which will probably vary between 10 and 50 per cent, depending on the machine capacity and gas density.

Thus, if the "gas losses" are estimated as 30 per cent, then the shaft horsepower is given by

$$\text{Shaft horsepower} = \frac{\text{THP} \times 1{\cdot}3}{\text{Mechanical efficiency}}$$

The mechanical efficiency varies between 88 and 92 per cent, depending on the machine loading.

Centrifugal Compressors

The determination of horsepower is based upon the calculation of head required for the compression, representing the work being done per lb of gas being handled, and is expressed in terms of feet (*i.e.* ft-lb/lb) as for a liquid pump. However, for a centrifugal compressor, where the specific volume is a variable, a more complex relationship obtains. Assuming the compression is polytropic, the fundamental equation for head is represented by:

$$H = z R_o T_1 \cdot \frac{n}{n-1} \left[R^{\frac{n-1}{n}} - 1 \right]$$

The value n is an exponent indicating the path of compression between suction and discharge and is greater than k, due to friction of the gas as it passes through the wheels and passages of the compressor when heat is added. Probably the easiest way to evaluate n is by the expression

$$\frac{n}{n-1} = \text{Efficiency} \times \frac{k}{k-1}$$

The polytropic efficiency is established by the manufacturer's tests, and for estimating purposes it can be assumed to lie between 70 and 78 per cent. The larger the compressor, the higher the efficiency.

The compressibility factor deviates from unity for real gas compression, and in most instances varies between 1·02 and 0·9. Thus for a general approximation the value z may be omitted.

The power required for the compression of a gas may be calculated from the following expression:

$$\text{Gas horsepower} = \frac{WH}{33000 \cdot E}$$

The compressor shaft horsepower is equal to the gas horsepower divided by the mechanical efficiency. For most applications the mechanical losses are small, and an average efficiency of 99 per cent may be used for estimating purposes.

When the discharge temperature for an uncooled compression exceeds 450° F internal diaphragm cooling may be utilized. In such instances the average exponent of compression is approximated by the isentropic exponent, and the head may be estimated on this basis.

Centrifugal Pumps

For a liquid pump, where the specific volume or density is constant, the fundamental equation for head becomes

$$H = \frac{144(P_2 - P_1)}{\text{Density}}$$

$$\text{Hydraulic hp} = \frac{W \times H}{33000} = \frac{\text{U.S. gal/min} \times \Delta P \text{ (psi)}}{1715}$$

Reference to standard curves will give the basic efficiency, and by applying a viscosity correction factor, if applicable, the overall efficiency (E) is obtained.

$$\text{Then bhp} = \frac{\text{Hydraulic hp}}{E}$$

Electric Motors

When sizing motors it is suggested that the hp rating be dictated by the following considerations.

For centrifugal pumps, use 110 per cent design bhp or 90 per cent of end of curve bhp, whichever is the less, followed by the nearest standard motor frame size up to about 200 hp, above which motors are normally custom-built.

For reciprocating pumps and compressors, use design bhp plus 10 per cent, followed by the nearest frame size.

The kW input to a motor is given by the relationship

$$\text{kW} = \frac{\text{bhp} \times 0{\cdot}746}{\text{Motor efficiency}}$$

The efficiencies quoted by manufacturers are for full load running conditions, and should be shaded down when calculating normal operation kW consumptions.

Steam Turbines

Approximate steam rates for single-stage back pressure turbines at various wheel diameters and up to about 400 hp are calculated from manufacturers' curves. Such curves normally cover a range of speeds and demonstrate that the specific steam rate increases considerably with decrease in speed. Referring to a normal speed of 3000 rev/min, rates for 6000 rev/min will be of the order of 75 per cent and for 1500 rev/min, rates will increase to 175 per cent or so. Above the 400-hp range and for large condensing turbines, individual steam consumptions may be appreciable, and accurate data should be obtained from vendors.

Diesel Engines

Figures for specific fuel consumptions will be available from manufacturers' literature, as will starting air, jacket water, and oil cooling consumptions. Full load fuel consumptions will be of the order of 7000 Btu/bhp/hr for large units.

Gas Turbines

The gas turbine produces two forms of power—shaft horsepower output and heat energy which can be recovered in the form of exhaust gas. The thermal efficiency of a simple cycle gas turbine is around 18 per cent based on shaft horsepower alone, but with a regenerative cycle a fuel saving of as much as 30 per cent at rated load can be expected and even greater savings at part loads.

For applications where exhaust heat recovery is an economic proposition, there are various means existing for utilizing the waste heat to increase the thermal efficiency of the cycle to the 40–70 per cent level, depending on the final exhaust temperature. Such methods include the use of the gas turbine exhaust to heat feed water for steam plants, to generate steam in unfired waste heat boilers, and as highly preheated combustion air for conventional steam boilers or for process use.

Typical energy requirements for a 7000-hp gas turbine operating alone and in combination with some form of waste heat recovery are indicated in Table II, where energy chargeable to the turbine and to steam raising are shown separately.

Actual performance data for the gas turbine can be obtained from manufacturers, and such information includes weight flow and temperature of the turbine exhaust, fuel consumption, and the turbine output horsepower which permits the thermal energy recoverable in the exhaust gases to be calculated. The cycle efficiency is defined as the useful energy output (mechanical work plus heat absorbed) divided by the total heat input.

TABLE II

Description	Btu/hp/hr		
	Heat input	To gas turbines	To steam
Simple cycle gas turbine exhausting to atmosphere	14,500	14,500	—
Regenerative cycle gas turbine exhausting to atmosphere	10,500	10,500	—
Simple cycle gas turbine with exhaust heat recovery boilers and varying exit stack gas temperatures	14,500	7,800–10,200	6,700–4,300
Single cycle gas turbine and steam boiler using exhaust gas as combustion air	14,500	5,900	8,600

THE FUNCTIONAL "UTILITY PREDICTION" CHART

The chart (Fig 1) illustrates the functional type of organization in which, after release of process and project design data, the subsequent mechanical analytical work is carried out by specialist personnel covering particular aspects of design. This is exemplified by the titles of the specialist divisions which form part of the Design Engineering Department.

The process, project, and engineering departments are, of course, responsible for engineering design and drafting work at all stages of a project; however, for the purpose of this study, their functions are considered only from the point of view of utility build-up and prediction during the estimate or pre-budget appraisal stage.

WORK BY PROCESS ENGINEERING DEPARTMENT

Process Design Information

The basic data comprise the process flowsheet, pump and compressor operating conditions, exchanger and furnace load sheets, vessel sketches, and process description.

Information on the flowsheet will include non-recoverable process requirements for steam, water, air, and inert gas, also flow and thermal duties for normal and design conditions.

Before general process release, the preliminary utility design basis will be discussed and agreed with the

Project Department. The review will cover such points as the re-use of steam and water, air *v.* water cooling, and, particularly, driver selection, and at the same time ensure that the various process considerations which directly influence utility consumptions and which have been previously discussed have been fully investigated.

WORK BY PROJECT DEPARTMENT

The Basic Project Information

This comprises the Project Design Data and Classification of Accounts, which are distributed to the specialist divisions at the same time as the general process release.

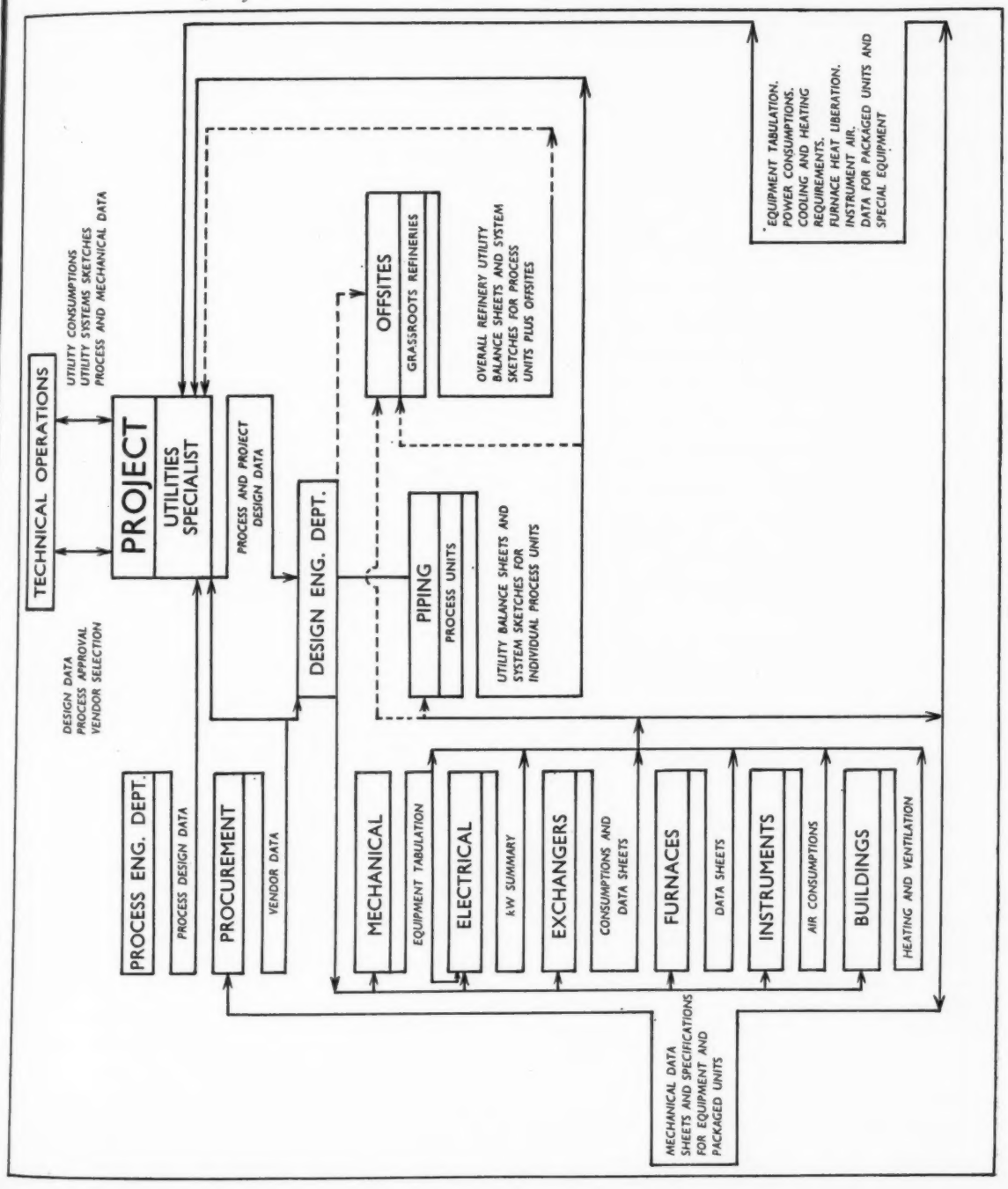

FIG 1
FUNCTIONAL UTILITY PREDICTION CHART

The Project Design Data list all known site information about utility services, establishing the quality, availability, and cost of existing steam, power, water, and air systems, fuel types with calorific value, and meteorological and other data.

The Classification of Accounts lists all equipment against item numbers and selected types of main and stand-by drivers. This document is subject to revision as the overall steam-power balance is developed and is at all times the basis for listing equipment on utility balance sheets. Revision may also include such items as inert gas generators, condensate flashpots, starting air compressors, and emergency equipment which could not be predicted in the early stages.

WORK BY SPECIALIST DIVISIONS

Mechanical Equipment

Based on design operating conditions, preliminary figures are calculated for pump and compressor bhp, motor ratings, steam and water consumptions for turbines, engines, surface condensers, and special equipment, and are recorded on the Equipment Tabulation together with recommended vendor selection and other information.

Design values of flow and pressure differential will be sufficiently accurate to calculate bhp, fix frame sizes, and to estimate approximate overall power, steam, and cooling water loads. Depending upon the final accuracy required, data sheets and specifications are sent to manufacturers for large compressors, pump, drivers, and special equipment as required to supplement existing typical performance curves and cost information. The data so received are used to check the calculated values.

The method for predicting the normal consumptions by reference to flowsheets and the equipment tabulation is demonstrated in the worked example, but for important users the problem should be discussed in detail with several manufacturers and normal power or steam requirements taken directly off the performance curves for the selected drivers.

Condensers, Coolers, and Reboilers

An " order of magnitude " figure for cooling water consumption is required initially for sizing mains and offsite installations, and to this end a preliminary assessment is made listing all consumers and estimated cooling water temperature spreads for each item.

The exchangers will be sized for dealing with design quantities and for the governing case, and, given the temperature spreads, the individual unit consumptions can be calculated for normal duties from flowsheet thermal loads. This is perfectly satisfactory for the preliminary assessment.

The summation of these normal flows will indicate the " once-through " utilities required to operate the unit. However, if the coolant is recirculated the requirements for clear make-up water will be calculated and recorded separately.

Similarly, the steam flow for other than the exchanger design case can be calculated from the heat loads shown on the process flowsheet.

Electrical Equipment

Normal and design bhp values for all motor drives taken from the Equipment Tabulation are used as a basis for the kW summary. Estimated motor efficiencies are applied to compute the kW input to each machine, and allowance is made for reduction gear losses and intermittent or fractional operation when applicable.

The kW summaries are checked directly by Project Department against bhp and efficiency values from vendor's performance curves when these have been obtained by the Purchasing Department.

Lighting and instrument loads are assessed and included in the summary.

Furnaces

The furnace type and its efficiency are assessed from the process design data, and the heat liberation and fuel consumption computed from heat duties shown on the process flowsheet. Efficiencies for packaged units will be supplied by the manufacturer. Emergency and decoking steam and air requirements are estimated for inclusion in the balance sheets.

Atomizing steam requirements for use with oil-fired burners are normally assessed at 0·2 to 0·3 lb of steam/lb of fuel oil, depending on viscosity.

Instrument Air

The normal air requirements for a pneumatic control system are calculated on the basis of $\frac{3}{4}$ SCFM for each control loop comprising transmitter and controller and $\frac{1}{2}$ SCFM for each pneumatic pilot and valve positioner. A factor of 20 per cent is added to the total normal requirements to arrive at a figure for design purposes.

Miscellaneous Intermittent Users

(1) *Space Heating.* The heating requirements for buildings will vary with the type of construction, inside and outside temperatures, window and door areas, and the number of air changes and can be calculated; but initially steam quantities per 1000 cu ft of space to be heated are assumed, *e.g.*

Type of building	Assumed air changes per hour	Steam, lb/hr
Administrative buildings .	4	10 (max)
Control houses . . .	18	10 (min)
Industrial buildings (sheeted)	15	20–30

(2) *Tank Heating.* Steam for tank heating is required to maintain the temperature of the tank contents for a number of reasons. Foremost of these is to lower viscosity and reduce pumping costs, prevent separation of liquids, avoid crystallization or condensation, and break emulsions. Heating steam can amount to as much as 20 per cent of the total steam requirements, depending on the proportion of black oil and bitumen storage and the degree of severity of the winter ambient temperature.

The heat lost to the surroundings only will be considered, and this is made up of the heat lost through the tank walls and roof for insulated and uninsulated storage.

Assuming the tanks are full, the heat lost through the wall and roof of a tank can be calculated from the formula

$$Q = U \times A \times \Delta T$$

where Q = heat lost, Btu/hr;
A = area of wall or roof, sq ft;
ΔT = difference between storage and ambient temperatures, ° F;
U = heat transfer coefficient, depending on the viscosity–specific gravity ratio; average values of U can be assumed as 2 for uninsulated walls, 0·4 for insulated walls, and 0·86 for roofs in all tanks.

The steam requirements can be calculated from the summation of the heat losses in walls and roof resulting from the above equation.

For design purposes the tanks will be assumed full and subjected to winter conditions, but for normal consumptions diversification factors will be applied to cover the fact that all storage will not be full at one time, and that less steam will be required during the summer.

(3) *Line Heating.* Arbitrary figures are used in the preliminary survey to assess steam requirements for line tracing.

(4) *Fire Fighting.* In a refinery it is customary to set the maximum fire water requirements at the probable location of the most dangerous fire and to assume that only one fire will occur at any one time. Water quantities are calculated for a specified rate of foam production plus additional water to cool the adjacent equipment.

For a process unit, an arbitrary figure of 500 to 750 U.S. gal/min is allowed for each unit.

Utility Balance Sheets

(1) *Process Units.* The balance sheets are prepared for each utility, and are intended to provide a complete mass balance in which the source, user, and destination of all material is given. They summarize all utility users and indicate the producers, consumers, and the net product or consumption of the unit as a whole.

The summaries are prepared as early as possible from data received from specialist divisions for use in utility pipe sizing, and are kept up-to-date with process and equipment sheet revisions. They record design flows for all users, including intermittent and emergency requirements.

Normal flows, when recorded, will be subject to review when information has been finalized and the estimate is nearing completion.

The general style of the balance sheets is shown in the worked example which follows, but obviously no standard form is possible, and the layouts will be adapted to suit the current problem.

(2) *Grassroots Refineries.* The net utility balances for all process units within a refinery complex are correlated and overall process requirements predicted, taking into account the expected operating and shutdown programme as influencing large intermittent steam, air, and water requirements for regeneration, decoking, and other duties.

In addition, there will be the supply to the offside installations which are situated outside process unit battery limits and which include such items as tank farms, loading facilities, administrative area, the central boiler and power houses, cooling water and effluent systems, also sulphur recovery and similar units.

For calculation of normal consumptions, diversification factors will be applied to cover intermittent pumping, product loading, and seasonal variations affecting tank, pipe, and space heating.

Finally, a balance is prepared covering all utilities and pressure levels to arrive at an overall optimum design for sizing the central utility installations.

System Sketches and Diagrams

A simple form of line diagram is used to illustrate flow and balance in individual systems when it is required to draw up overall utility balances for a process unit or refinery.

The diagram for the steam system in the crude unit example is illustrated in Fig 2. Here the instrument positions have been indicated, enabling any deficiencies in measuring instrumentation to be recognized and corrected.

The steam system in Fig 3 is more complex, and comprises a main boiler plant feeding one process unit and three large turbines whose steam consumptions can be accurately assessed, for normal and design conditions, from the manufacturers' performance curves.

Both the 200- and 40-psig mains are fed from back pressure turbines to supply steam requirements for the process units and offsites as calculated on the utility balance sheets. The balance of the 200-psig steam is sent to a mixed pressure condensing turbine under manual control, which can be varied to allow for heavy intermittent and emergency loading.

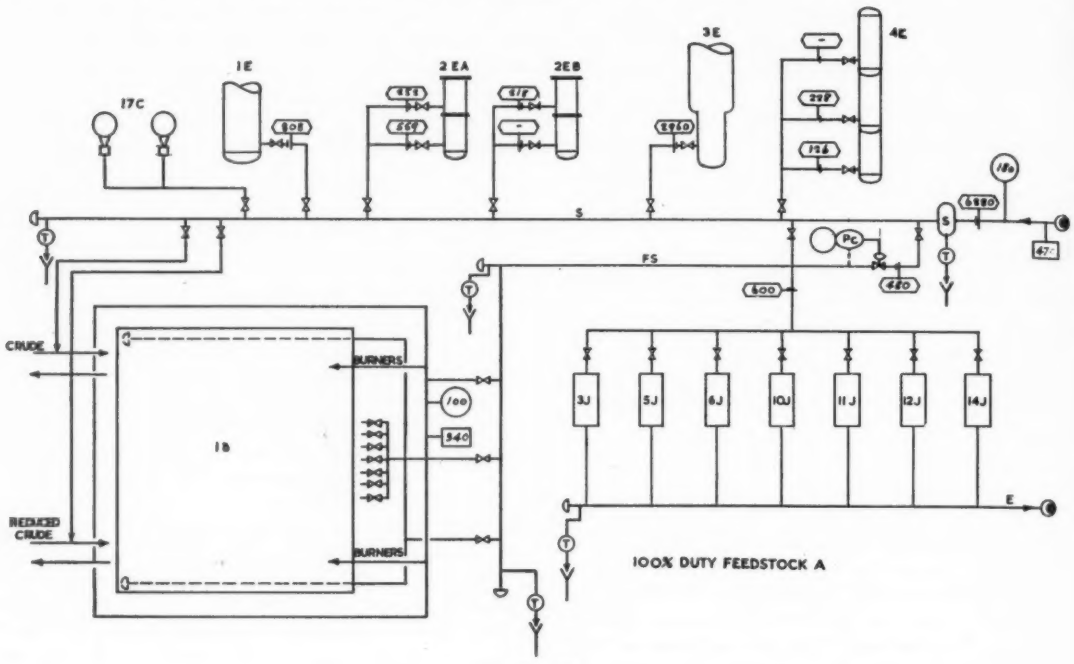

FIG 2

BLOCK DIAGRAM—STEAM SYSTEMS

The system is in balance theoretically, but additional flexibility is afforded by pressure-reducing

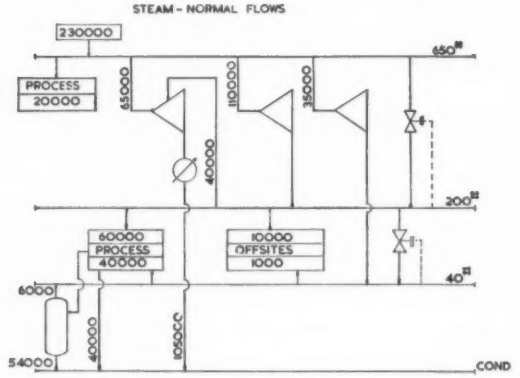

FIG 3

stations and, possibly, by the addition of stand-by turbines not shown on the diagram.

The main boiler plant is then sized for the high pressure steam requirements under design conditions.

THE ESTIMATED UTILITY SUMMARY

The prediction of normal utility values and the preparation of the final summaries covering all operational cases is the responsibility of the Project.

As has been seen, the primary function of the specialist divisions is to provide design quantities from which equipment, piping, and services can be sized, and estimated space requirements, so that the overall layout of equipment can be finalized, but normal utility flows will also have been predicted and recorded on the balance sheets to assist in sizing the central boiler and power generation plants and cooling water system. However, it is left to Project, and particularly the Utility Specialist, to assess these quantities accurately along the lines to be discussed.

Project will probably use the documents and balance sheets prepared by the specialist divisions as a basis for checking, as these will have been under review during the systematic build-up of the estimate and will supply the latest overall picture in a convenient form.

The work to be undertaken by Project during the final survey and in preparation of the predicted totals can be summarized as follows:

1. Check that all equipment listed on the balance sheets conforms to the latest equipment lists for process units and offsites.

2. Check that factors set by the process data have been applied in calculating design values by specialist divisions and that allowance has been made to cover intermittent and fractional loading of equipment from whatever cause, and check diversification factors which have been used to

cover seasonal requirements in the offsite installations.

3. Amend the calculated bhp values, specific consumptions for drivers, and equipment efficiencies by reference to data and performance curves obtained from vendors, and check that utility requirements for packaged or specialized units sold under licence conform to the vendor's published data with special reference to any guarantees.

4. For the remaining equipment, calculate figures for straight or multi-case operation, based on design data supplied by specialist divisions, using normal values for each case taken from the flowsheets. This involves a review of equipment efficiencies, cooling temperature spreads, and other factors which may be influenced if the normal differs appreciably (more than 10 per cent) from the design values. This aspect is illustrated in the worked problem, where pumps and large exchangers were sized for a future double capacity unit.

5. Check that the electric power consumptions conform to the revised bhp values and predict the final total loads.

6. By reference to the utility check list (Table I) ensure that all services listed under the three categories have been considered.

7. Finally, complete and check the balance sheets and prepare the final summaries of utility consumption for all operational cases, treating separately each process unit and the offsite installations.

WHAT OF THE FUTURE?

The formation of a specialist group for handling utilities will undoubtedly lead to closer predictions in the future by the canalizing of information and the comparison of predicted quantities with those achieved by operators on site.

The recording of utility data at start-up of a process unit tends to be neglected, partly because operators are primarily occupied in ensuring that a unit fulfils its process guarantees, and partly because of deficiencies in measuring instrumentation on utility lines and equipment which usually exist when no systematic records or guarantees are envisaged during the design.

Instrumentation should be reviewed at an early stage in the design to ensure that it is adequate to measure the quantity and quality of all utilities in and out of battery limits; also at major individual consumers and producers where the resulting data are considered to be of sufficient importance to justify the slight increase in capital cost involved.

The importance of comparing the predicted and achieved figures, whenever possible, cannot be overstressed, as by so doing a fund of information will be built up which can only lead to a better understanding of the problem and tend towards greater accuracy in the design and prediction of utilities.

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APPENDIX

A CRUDE DISTILLATION UNIT PROBLEM

To illustrate in detail some of the basic calculations and assumptions, the balance sheets, and other features of utility prediction, a hypothetical atmospheric/vacuum crude distillation unit has been used as a basis for reference.

The unit has been designed for operation at 50 per cent duty for an unknown period and handles two different feedstocks.

The tabulated utility consumptions are assessed for all cases and are confined to the actual expected requirements to operate the unit and, as such, would form the basis of any contractural guarantees.

The following documents have been included to illustrate the problem:

Check list of utility users
Power consumption
Balance sheets for steam, cooling water, and fuel
The summary of utility consumptions

Estimated Utilities Summary

	Normal			
	100% duty		50% duty	
	Case A	Case B	Case A	Case B
Steam				
A. 180 *psig*, 470° *F*				
Consumption:				
Process, lb/hr	5800	5470	2900	2735
Drivers, lb/hr	600	1300	290	650
Atomizing, lb/hr	480	450	240	225
Total, lb/hr	6880	7220	3430	3610
B. 15 *psig*, *D and S*				
Production:				
Drivers, lb/hr	600	1300	290	650
Cooling water, in 75° F				
Consumption, U.S. gal/min	1615	1685	760	790
Electric power, 420 volts				
Consumption:				
Motors, kW	108	109	69·5	70
Lighting and instruments, kW	25·5	25·5	25·5	25·5
Total, kW	133·5	134·5	95	95·5
Fuel oil LCV, 17,200 Btu/lb				
Consumption, lb/hr	1600	1490	800	745
Air, 80 psig				
Consumption:				
Instruments, SCFM	30	30	30	30

Check List of Utility Users

P = Producer C = Consumer

Item no.	Description	Steam 185 psig	Steam 15 psig	Cooling water	Steam cond.	Fuel oil	Hot water return	Electric power
1B	Furnace	C				C		
7C	Condensers			C			P	
8C				C			P	
9C				C			P	
10C				C			P	
11C	Coolers			C			P	
12C				C			P	
13C				C			P	
14C	Cooler boxes						C	
15C							C	
16C							C	
18C							C	
17C	Vac condenser	C		C	P		P	
1E		C			P			
2EA/B	Towers	C			P			
3E		C			P			
4E		C			P			
1J								C
2J								C
3J		C	P	C				
4J				C				C
5J		C	P	C				
6J		C	P	C				
7J	Pumps			C				C
8J				C				C
9J				C				C
10J		C	P	C				
11J		C	P	C				
12J		C	P	C				
13J				C				C
14J		C	P	C				
15J	Compressor	C	P	C				

Electrical Power Consumptions

Item no.	100 per cent duty						50 per cent duty		
	Rated hp	Motor efficiency	Design bhp	Normal bhp	kW input A	kW input B	Normal bhp	kW input A	kW input B
1J	75	0·92	68	51	41·5		33	27·4	
				51		41·5	33		27·4
2J	10	0·88	8·6	7	5·9		4·6	4	
				6·5		5·5	4·2		3·6
4J	5	0·85	3·9	3·2	2·8		2·1	1·9	
				2·4		2·1	1·7		1·5
7J	10	0·88	8	7·1	6		4·3	3·7	
				7·5		6·3	4·6		4
8J	15	0·88	12·2	9·8	8·3		5·8	5	
				12·1		10·2	7·2		6·2
9J	60	0·91	49·5	47	38·5		28·4	23·8	
				45		36·9	27		22·6
13J	40	0·90	30	6·1	5·1		4·2	3·6	
				7·8		6·5	5·5		4·7
15J	40	0·90		(28)			(28)		
Total load motors, kW					108·1	109·0		69·4	70·0
Lighting load, kW					23	23		23	23
Instrument load, kW					2·5	2·5		2·5	2·5
Total plant load, kW					133·6	134·5		94·9	95·5

Notes: 1. Normal bhp are based on flowsheet quantities with assumed pump efficiencies at 100 per cent and 50 per cent flow duties.
2. All pumps are sized for 100 per cent duty.
3. Intermittent consumptions are in brackets.

Balance Sheet for Fuel

The furnace has been sized for 100 per cent duty, but tubes and burners have been omitted initially for 50 per cent duty.

100 per cent duty

Item no.	Heat duty	Heat liberation	100% oil firing, lb/hr			Atomizing steam, lb/hr		
	MM Btu/hr		Design	Normal		Design	Normal	
				A	B		A	B
1B	20·705	27·6	1600			480		
	20·705	27·6		1600			480	
	19·224	25·6			1490			447
	Total oil		1600	1600	1490			
	Total atomizing steam					480	480	447
	Total including intermittent and stand-by		1600	1600	1490	480	480	447

Note: For 50 per cent duty, halve the above quantities. Assume furnace efficiency is 75 per cent and fuel oil LCV = 17,200 Btu/lb.

Utility Balance—Steam

Item no.		Steam flow, lb/hr, 180 psig 470° F				Condensate, lb/hr			
		Crude A		Crude B		To 15 psig steam		To sewer	
		Normal	Design	Normal	Design	Max normal	Design	Max normal	Design
	100% duty (flowsheet quantities)								
1E	Crude tower	808	808	1070	1070				
2EA	Stripper:								
	No. 1	252	250	197	347				
	No. 2	559	559	406	805				
2EB	Stripper:								
	No. 1	218	218	119	120				
	No. 2	—	158	—	158				
3E	Vac tower	2960	2960	2680	2680				
4E	Stripper:								
	No. 1	—	247	—	247				
	No. 2	228	228	329	329				
	No. 3	126	126	119	119				
17C	Vac condenser	650	650	550	550				
	Process steam (total)	5800	6205	5470	6425			5800	6425
	Drivers	600	900	1300	1600	1300	1600		
	Oil burners	480	600	450	600				
	Total steam	6880	7705	7220	8625				
	To LP steam					1300	1600		
	To sewer							5800	6425
	Total including intermittent users	7080	7705	7420	8625	1500	1600	5800	6425
	50% duty (flowsheet quantities halved)								
	Process steam	2900	3100	2735	3210				
	Drivers	290	450	650	800				
	Oil burners	240	300	225	300				
	Total steam	3430	3850	3610	4310				
	Total including intermittent users	3530	3850	3810	4310	750	800	2900	3210

Unity Balance—Cooling Water

Item no.	Description	Δt, °F	Thermal duty design	Cooling water					U.S. gal/min	
				Inlet (100% duty)			Inlet (50% duty)		Hot return design	To sewer design
				Design	Crude A	Crude B	Crude A	Crude B		
					Normal		Normal			
7C	S and T condenser	30/35	5·54	369	320	369	140	160		
13C	S and T condenser	30/35	9·63	643	599	643	255	275		
8C		20	0·60	60	17	21	8	11		
9C	Double-pipe	35	0·83	48	19	25	10	13	1265	
10C	Coolers	35	1·48	85	28	42	14	21		
11C		30	0·30	20	7	20	3	10		
12C		35	0·71	40	—	—	—	—		
	Cooler boxes	26	4·96	—	—	—			−380	380
Inlet	Exchangers		24·05	1265	990	1120	430	490	885	
	Vacuum condenser	15	4·70	620	560	500	280	250	620	
	Pump cooling			95	65	65	50	50		95
	Total U.S. gal/min			1980	1615	1685	760	790		
Thermal duty, MM Btu/hr			28·75							
Hot return, U.S. gal/min									1505	
To sewer, U.S. gal/min										475

Notes: 1. Design inlet temperature of river water is 75° F.
2. Exit water from exchangers at 110° F is re-used in cooler boxes and discharged to sewer.
3. Exchangers 7C and 13C are sized for 100 per cent duty as use of 50 per cent units would result in sub-standard dimensions.

DISCUSSION

D. P. Plummer (Shell Refining Co. Ltd): Utilities have to be absolutely reliable, and as a consequence developments and inventions seem to be a little slow. Steam is raised in refineries to-day in much the same way as it was 20 years ago. But there is not much resemblance between the way in which petrol is made to-day and the method used 20 years ago, and for that reason utilities tend to be thought of as being a little less exciting. However, it is really impossible to over-estimate their importance, not only from the point of view of the refiner but also the choice of contractor, who will certainly be favoured if he is sure about his utilities. Plants are made under guarantee, and if 50 per cent has been added because the contractor is not quite sure, that will probably put the figure higher than that of a contractor who is certain of how much steam and heat will be required. Choice of process also comes into it. If there is more than one process that will do the job the refiner is very often influenced by the consumption of utilities. It is amazing how disinterested managers and buyers can be in the technical details of the process. They say, "What does it cost to run? How much steel do you want? How much this and that?" And on that often depends the decision. In fact, one can say more—that there have in fact been processes that one or two refiners have had to use because there has been no alternative, but they are very greedy with steam or fuel consumption. They are not popular until someone comes along and halves the steam consumption, and then it is suddenly found that everyone is building them.

The bane of utilities is that people think there is a sort of rule of thumb about them. They ring one up and say "Distilling units. What is a good figure, pounds of steam per ton of . . ." and if they are not given an answer they think one is ignorant. The author has made no attempt to give such a figure. I have been considering recently three quite different plants. One was to distil 10,000 tons a day of a certain kind of crude oil; another was to distil 3000 tons a day of another crude oil; and a third was to separate 1200 tons a day of *normal* and *iso*-pentane. And each consumed 180 million Btu/hr of heat. But it can be seen that the throughput was very different in each case.

The author has said and done the right things, he has calculated from each part, and from the parts made a summation to arrive at a total figure. There is no other way to give it.

I would like to have seen in the paper a little more help in choosing which form of utility to consume. For motive power one could have electricity, steam, or a diesel engine, and for heating one can either use a hot oil system, direct firing, or latent heat of steam, and there is a choice between them. I would have liked to see a little discussion of that in the paper, perhaps emphasizing the comparatively rare use of diesel engines for motive power.

With modern refineries one tends only to use steam if one can use its latent heat. The way in which the thing is often done is to find out how much steam one wants to use for process, and having obtained one's LP steam requirements, work backwards to the HP. Then, of course, everything that cannot be driven with that amount of high pressure steam is taken in the form of electricity. We are very often forced to doing that kind of thing.

F. P. Alban: It is of course very important, as I stressed in my paper, for investment costs of utilities to be kept down, as they comprise a very considerable proportion of a new refinery construction. As I said before, they can equal or exceed the total cost of the process units, and to achieve this minimum cost design we require, as early as possible, a reasonably accurate estimate of all the utility figures. The main consumers of utilities are undoubtedly the large drivers, whether actuated by electricity, steam, or gas, and the important thing is to study all possible alternatives on the basis of fixed plus running costs over a suitable period, with due regard for utility balances.

Each refinery site will present its own problems, which might include the existence, or otherwise, of reliable public utilities or natural sources of water or fuel, all of which will influence investment and running costs. Also, the refinery management may have distinct preferences, and all these factors will be considered in the design.

K. B. Barnes (*Oil and Gas Journal*): This is a most informative and beneficial report, and I have only one question to ask, concerning the matter of relating these predictions which, after all, deal with a refinery project of a definite size and capacity, to future growth and expansion. In almost all refinery operations the units start out at a certain charging rate and then, in two, three, or five years, the requirements have increased to some new and higher figure. My question therefore is how does one account or allow for future expansion in these calculations?

F. P. Alban: I have considered the case of a unit which would operate at 50 per cent duty for an unknown period, but which was sized for 100 per cent duty to allow for future expansion. There are several decisions that have to be made regarding the capital equipment. One of these is whether it is cheaper in the long run to install, in the case of pumps and compressors, three 50 per cent units to start with, one being a stand-by, or whether to put in two 100 per cent units right away. The question of space and investment cost will influence the decision, but from the point of view of running efficiency it is obviously more economical to put in the 50 per cent units, because the running of pumps at less than 50 per cent of their design capacity brings the efficiency down very considerably, possibly by 10 to 15 per cent, and that will, over a period, considerably increase the operational costs.

The same thing applies to exchangers. Normally, it would be advisable to use 50 per cent exchangers, and then add another unit later on. But this is not always possible when dealing with small units, as the 50 per cent job could result in sub-standard dimensions, in practice necessitating the use of exchangers sized for 100 per cent duty.

Furnaces are simple, because one can design the furnace shell to accept additional tubes at a future date for increased capacity, while towers and drums will normally be sized for the future duty.

W. H. Foster (Kellogg International Corporation): The paramount question, it seems, having been told how to assess the utilities with reasonable accuracy, is how much is the plant going to cost to run? I think that is the criterion for all operators, although it might not necessarily be the criterion for a designer. It seems that much of the wealth of knowledge of how much a plant costs to run rests firmly in the hands of refiners, who run these plants, and who, I feel, should be able to give much information on the relative costs of the different forms of power. I would suggest that a paper by a member of one of the oil companies is highly necessary so that the relative costs of utilities could be given on the basis of years of experience.

A point raised earlier by Mr Plummer, which is really outside the scope of the paper but cannot be easily dismissed, is what sort of prime mover will be chosen, and what are the conditions that dictate the kind of prime mover to be used? There is one aspect which has not been mentioned but which is of vital importance. In the case of electrical power, and by ramification whether a particular drive is steam or electric, the answer depends to a great extent on the tariff at which the electricity is purchased. One may find oneself in a situation, and this has actually happened, where in order to get a preferential tariff one may have to guarantee a minimum load. But one may well find that to benefit from that tariff it is advisable to use electric drive even when other conditions show it should be steam. The contractor is usually given the cost of the energy based on unit charge only at the equivalent of about a penny a unit. The maximum demand charges are rarely given, and their conversion to a unit charge is never supplied. I think it can be seen, when one considers that a unit or refinery taking about 10,000 kVA costs about £150,000 a year to run, that one cannot be far out with unit costs, *e.g.* the difference between a penny and 1·2 pence would be something like £20,000 higher or lower on the running costs. To me this would be vital information and should receive careful thought when it is shown or given in the data around which the design takes place.

On this question also of experience in running these units, little or nothing is known of intermittent loads, and an accurate knowledge of intermittent loads is necessary. It is a function of day and night pumping, amongst other things. Oddly enough, the intermittent loads are usually on the big drivers and some further knowledge should be made available.

Referring once again to Mr Plummer's earlier statement on the question of choice of drive, a great deal of thought is being given at the moment by various contractors, and presumably the oil companies as well, on whether the acknowledged methods of driving are the right ones. I think it is well known that the predominance at the moment and in the past has been on steam. I am not at all sure that this is the right answer. It is generally known that the capital costs of large electrical drives, and this is pretty well immaterial of the size, is considerably cheaper electrically than any other way, although above about 1400 or 1500 hp no one seems to be yet satisfied that the running cost of an electric drive is cheaper than any other. But there are two schools of opinion. One school seems to be definitely of the opinion that there are considerable savings to be made by electric driving, and the other school does not. That is reflected in the extremes that we seem to go to to-day, with the large refineries being built which are all-electric and other large refineries which are all-steam. There seems to be no compromise, and nobody seems to be in a position to say, "This compromise, with half steam, half electric, has proved to be the right one, and we shall continue on those lines," or "The next refinery that we build will be all-electric or all-steam."

F. P. Alban: The type of drive will depend, I think, largely on the local unit cost of the particular service. For instance, on the Continent of Europe and in the U.K., the cost of power will vary between 1 cent and 1·8 cents/kw, whereas in the U.S.A. the average is 0·8 cent, so that one might expect the Americans to favour electric drives. However, this is not necessarily the case, as the decision

on the type of drive will depend on many factors influencing the choice, such as the cost and availability of natural and manufactured fuels. The reliability of the source is also most important. For instance, we would not rely on a single high-tension line to supply a large 4000-hp compressor which might constitute a vital link in a process. We would want two independent power supplies, each probably operating at 50 per cent load with provision for automatic switching in the event of the failure of one, or alternatively, a stand-by machine with steam or gas drive, which can be expensive. Each job will have to be judged on its merits, and the question of tariffs, which, as Mr Foster said, are dependent on maximum demand, will be considered in the design and when calculating annual operating costs.

In a process which is designed for multi-case operation, *i.e.* for alternate feedstocks or alternate product specifications, and possibly for expansion at some future date, the material and heat quantities for fractional and intermittent loads can usually be taken directly off the flow sheets, and these quantities will provide the basis for the calculations as illustrated in the worked example. Other types of intermittent loads are emergency compressors for instrument service and regeneration air, emergency power generators, and the like, which will be listed in the Classification of Accounts and treated separately in the final utility summaries.

When considering the type of drive, the gas turbine, which, for certain applications, can be just as efficient and economical as either steam or electric drives, must be remembered. Among such applications are those in which one or more portions of the gas turbine's cycle can become an integral part of a complete process. Large volumes of high temperature gases passing through a throttle valve provide conditions which are very suitable for power recovery, providing the pressure is right, quite apart from the normal waste heat recovery boiler or CO boiler which is sometimes installed. Obviously the ideal arrangement is to be in energy balance within the plant, and if we can get, in a catalytic cracker, say, all or part of the regeneration air requirements from the waste energy in the flue gases, then we have achieved significant economies.

J. K. R. Seal (Esso Petroleum Co. Ltd): Could Mr Alban give some indication of the relative efficiency of power recovery from catalytic cracker regenerator flue gas via gas turbine and by CO boiler/steam turbine; and some indication of the relative cost and economics of those two systems?

F. P. Alban: The conventional gas turbine cycle consists of compression, combustion, and expansion, and when the gas turbine is integrated with a fluid catalytic cracker the regenerator serves as the combustion chamber and the flue gases are directly expanded in the turbines which drive the air compressor. Prototype power-recovery gas turbines, which have been in operation satisfactorily for some time, provide the full compressed air requirements for the process, and result in power balance. I believe that turbo-compressor machinery is available in the U.S.A. which can meet the requirements, and in terms of air capacity they are available in a range of sizes up to about 17,000 scfm. I understand that a turbine efficiency of 78–80 per cent and compressor efficiency of 75–77 per cent are required if power balance is to be achieved. A CO-burning boiler can be added to the turbine exhaust, as there will be still 5–10 vol per cent unburned CO in the gases, but supplementary firing will be required. This combination permits the recovery of about three or four times the useful mechanical work obtainable from an unfired flue gas boiler or CO boiler by itself.

Regarding investment costs, I have no specific information about the CO boiler, but it is a very expensive piece of equipment—I believe that £500,000 was a figure quoted recently. For a catalytic cracker regenerator capable of burning 35,000 lb/hr of coke, I understand that the installed cost of the separator and gas turbo-compressor equipment would be about £500,000, which is approximately the same as for the equivalent steam turbine-driven compressor of conventional design. However, in the latter case there will also be the cost of utilities to supply steam and cooling water, say another £300,000.

B. H. Gibbs (Hayward Tyler & Co. Ltd): I was interested to hear why Mr Alban chose big motors or turbines. The small-size electric motor is cheaper, but when one covers all the flameproof requirements, also special cabling and switchgear, which are quite expensive, I am not so sure that it is much cheaper than a steam turbine. One has in many places in an oil refining unit heat which must be there; much of it goes to waste, and much can be used as steam for driving machines. In many of these cases where one has these process factors going in, utilities are calculated for 50 or 60 per cent more power requirement than is actually going to be used, or expected to be used, for quite a long time. Steam-driven stand-by units can be used for quite a long time, speed-controlled to actual requirements, and they are much more economical than motors.

There is one factor against that, which has been mentioned, and that is the question of minimum load rates. I think it is rather iniquitous and tends to uneconomical running because unless one meets the maximum demand one has a high rate. There is always a tendency to keep the electrical rate up, even when running at a small load factor, because of this rate imposition. I have worked for the Kellogg Company on their factors on a number of occasions, and I often wonder if some of those factors between normal and design are not made to cover what very often goes with an oil refinery. When it has been in operation for about six months up goes the throughput by another 50 per cent and everybody is as happy as can be. That seems to be the sort of thing that goes on, and I noticed that Mr Alban talks about the comparison between 50 and 100 per cent operating capacity. I have been working on a plant of that nature recently, quite a small plant, and we worked out the economics of constant speed motor or steam turbine drives, and I think the latter are a really good proposition. We first selected the pumps and operating speeds or impeller diameters for the 50 per cent capacity for a year or two years, and then said how much it would cost to bring up the capacity to double by fitting larger duty impellers for motor drives. This cost was zero for steam turbine drives, because on small pumps the extra quantity is obtained by a few per cent speed increase.

I take it that this is what is meant in the paper when Mr Alban says that one puts the pumps in initially for the 50 per cent rate and then alters them afterwards for the 100 per cent rate.

There is a terrific lot of wastage with these process factors if they are overdone when put into these considerations. If one is putting in pumps for 50 per cent more than the actual requirements, then a lot of wastage power is thrown away in the valves. There are means of providing margins and ways of recovering the power losses, particularly that in which we have been interested lately, by automatic or hand speed control of important units, instead of valve control, across a very wide range

of volume and differential pressure, even with constant speed ac motor drive.

F. P. Alban: I know that the general tendency is to oversize equipment when designing refinery utilities. As Mr Gibbs said, it is very nice to get more for the same price, or what would appear to be the same price. In actual fact it is not so, because if units are sized up, as Mr Gibbs said might happen, then generally speaking the price goes up with it. And if one is a contractor it is getting more and more difficult to obtain jobs if one oversizes. Especially is this true when dealing with Continental refineries, which are very cautious about the last penny when it comes to operating and investment costs, and it is our experience time and again that if we are to be competitive, then we have to design down to the proper safety factors.

Regarding the remark about the size of the pumps for 50 and 100 per cent, all the pumps in this particular example are very small, and the possibilities are that the same frame size should be used for both 50 and 100 per cent. There is just the question of changing the internals. I do not know whether that is so, but for pumps requiring about 20 hp I think you could do that.

B. H. Gibbs: In the case I worked on, one could barely make the pumps any smaller. The capacity is very small, and the smallest pump made will meet the initial half capacity and final full capacity with the same pump size.

F. P. Alban: Generally speaking, I think that if there is a possibility of operating at 50 per cent capacity for a long time, obviously the equipment should be sized properly at the beginning and 50 per cent units added later.

R. D. Hammond (Esso Petroleum Co. Ltd): First I would like to make the point that very often the choice of driver is determined by safety or other considerations, rather than by the actual cost of unit power.

Secondly, I would like to ask why safety facilities have been omitted from a paper on the systematic prediction of utilities. The assessment of these can be quite a headache at the preliminary stage of a project.

By safety facilities I mean plant on-site safety facilities, plus the associated off-site facilities, *e.g.* flare lines, flares, etc.

F. P. Alban: I have not dealt with that side of the question in any detail, because in a paper of this length it was necessary to restrict the subject matter. The water requirements for fire fighting are always predicted at the same time as the tank-heating and space-heating requirements for buildings. They are not included in the overall summaries, because after all they are not the utilities that are required to operate the plant. However, they obviously have to be considered to size pipelines and the main boiler plant. In a prediction of this sort we would consider the size of the area and estimate that so many hydrants would be required and a certain size of main. From a knowledge of the water pressure one can calculate the amount of water. But perhaps the problem is rather different with tank farms and areas where large volumes of foam are required at short notice. For instance, I think normally one inch of foam will be required over a certain area in ten minutes or so, and by working back we can get a pretty good idea of the maximum water that we are going to need. I have not dealt with blowdown because that is rather outside the scope of this paper.

G. C. Scott (The Distillers Company Ltd): I am very interested in the question of sizing. From previous remarks it would appear that in the case of smaller units, such as small fans and pumps, size is not of major importance. On larger utilities it is.

Consider the case of a new process which requires, say, 15,000 lb/hr of steam, of which there is already available 8000 lb/hr. Is one to install a new boiler plant having a capacity of 7000 lb/hr when one's standard boiler size is in the order of 30,000 lb/hr? Additionally, what size of steam main should be installed when the user is placed on an open site beyond which is space in which the factory will naturally expand? How far is one justified in exceeding the net minimum requirements of the utility for the quantity required for this plant?

F. P. Alban: Depending on your estimate of future commitments, I think one should install the minimum standard size boiler to meet one's requirements, say, 15,000 lb/hr in this case. However, if the proposed future expansion is likely to warrant it, perhaps one should consider the installation of a further 30,000 lb/hr unit, as this is the standard size.

G. C. Scott: There really is no absolute answer to the question. Nevertheless, a decision was made.

F. P. Alban: What was the decision, to install a small boiler?

G. C. Scott: No, to put in a 50,000 lb/hr unit.

F. P. Alban: I would say it would largely depend on the condition of one's existing boilers and whether capital was available for that other boiler. But I think the decision was probably correct if the future expansion is to be considerable.

J. M. Wentworth (Kellogg International Corporation): I would like to make a point not mentioned in this discussion so far. It is sometimes the case that the accountants are responsible for the choice of utilities, rather than the engineers. We do get cases in which the utilities specified to be used on certain sites are not the cheapest by any means.

F. P. Alban: I am afraid I cannot comment on that any further. I think I entirely agree with you. If it is taken out of the engineers' hands, what can we do, except to put up a good case to the accountants?

S. Vouyoucalos (Birmingham University): I would like to know whether, in the different data you obtain from the manufacturers, you use them so that you employ one set of machines of one make, or do you merely use the ones that give the best results? If, in the case of big-size pumps, one firm gives you better results, would you go to other firms for the small size? Do you prefer to have one from the same firm, so that you get better service later, or do you go to the ones that give individually best results?

F. P. Alban: I assume that your question relates to decisions to be made during the estimating stage and not when the job has been placed, when we have to make a final decision on the maker and type of unit, say a large compressor. Assuming that we are still in the estimate stage, the stage we have been dealing with throughout this talk, we would probably go to three or four manufacturers, on the Continent, in England, and in America,

and we would know quite a bit about them before we started. We would know the reliability of their products, and how they were costwise, more or less. For an important piece of equipment, such as a large compressor and driver, we would make an economic study covering fixed costs and running costs over a period based on cost and performance data from each manufacturer. The result of this study, plus such factors as known vendor reliability, after-sales service history, and customer preference, would determine the final choice. The choice of manufacturer in the case of smaller equipment will often be dictated solely by the need for standardizing certain makes of equipment within a refinery, which eases the spares and maintenance problems.

The Chairman then called for a vote of thanks to the author, which was accorded with acclamation.

THE HYDRODYNAMIC EFFECT BETWEEN APPROACHING SURFACES—ITS INFLUENCE ON SURFACE PROFILES DURING PLASTIC DEFORMATION *

By L. H. BUTLER † (Fellow)

SUMMARY

Compression tests have been carried out on 1-inch-diameter × 1-inch long cylindrical, mild steel specimens using flat contact dies. The die-to-metal interfaces were both unlubricated and lubricated with high viscosity fluids, and the moving die approach speed was controlled at 1 inch/min and 9 inches/min. The specimen contact face profiles were measured before and after compression. The final profiles and surface appearances indicate that under moderate approach conditions and using an appropriate lubricant, it is possible to deform the material well into the plastic range via a continuous lubricant film stimulated initially by the hydrodynamic mechanism. The film is maintained by a mechanical sealing effect after initial metal contact occurs. The consequences of the presence of such films is discussed with reference to the magnitude and constancy of values of coefficient of friction during forming operations.

INTRODUCTION

THERE is little reason to suppose that during bulk metal plastic forming operations (rolling, forging, wire drawing, etc.) surface contact and friction conditions are identical to those suggested by Bowden and Tabor [1] for bulk elastic contact. Attempts to measure values of coefficient of friction [2] during such operations suggest that they are lower than might be expected from conventional sliding tests. Also, observations of the surface finish of deformed materials [2, 3] show effects which indicate that in many cases surface disruption has occurred during deformation in the absence of die or tool to metal contact.

In a previous paper [4] an analysis and discussion of the hydrodynamic effect between normally approaching surfaces suggested that even under moderate approach conditions continuous lubricant films might be established with pressure distributions sufficient to initiate plastic yielding before metal contact occurred. The film thicknesses, although small, would be of a sufficient magnitude to exceed the asperity heights of surfaces encountered in normal engineering practice. The pressure distribution was such as to induce dishing of the deforming surface maintaining a trapped lubricant pool throughout subsequent deformation. The presence of this lubricant film would satisfactorily account for the observed surface deteriorations.

The present work supplements the previous theoretical analysis and describes some experiments which have been conducted in an attempt to measure any change of surface profile after deformation, which can be attributed to the hydrodynamic effect.

MATERIALS USED

Specimens

1-inch diameter × 1-inch long cylindrical mild steel specimens (approx 0·2 per cent C, with a tensile strength of 32·24 tons/sq in), were machined from $1\frac{3}{8}$-inch diameter bar, previously annealed at 900° C. The ends of the specimens were fine turned, and sometimes polished, to be as accurately flat and perpendicular to the cylindrical axes as possible.

Lubricants

Pure paraffinic base oils of high viscosity were used as follows:

Lubricant No. 22: viscosity—45·4 cP at 210° F
Lubricant No. 19: viscosity—756 cP at 100° F

The lubricants, of which No. 19 is the less viscous, and their details were supplied by C. C. Wakefield & Co. Ltd.

Compression Dies

The bottom (stationary) die was a 2-inch diameter flat, hardened steel disk with no special surface preparation. The top (moving) die was a $1\frac{1}{2}$-inch diameter × $1\frac{1}{2}$-inch long hardened steel roller with one of its flat end faces (the contact face) polished and flat to within 0·00005 inch maximum.

TEST PROCEDURE AND SCHEDULE

Each specimen was set on a surface plate and one end face profile was measured with a dial gauge across three diameters A, B, and C at 120° to each other.

* MS received 1 April 1960.

† University College, Cardiff.

Gauge readings to 0·00001 inch could be estimated, and the same part of the surface plate was used throughout to reduce the effect of any inaccuracies in its profile. All readings were rechecked throughout and were found repeatable to within 0·00005 inch maximum.

The specimens were degreased in acetone and placed in turn on the bottom die with the contact (measured) face upwards. The selected lubricant was applied liberally to this face and the top die placed in contact with it. The load was applied to the top die through a previously aligned spherical seating carried in the top crosshead of the universal testing machine being used. The approach velocity of the crosshead was controlled at either 1 inch/min or 9 inches/min as required, but it should be borne in mind that these were the velocities before contact. At the instant of contact and thereafter the strain rate was unknown, as in a hydraulically operated machine, such as the one used, the strain rate varies with the load.

Compression was continued until the specimens became approximately 0·9 inch long, when they were removed, cleaned, and the top face profiles remeasured and inspected.

The schedule of tests is given in Table I.

TABLE I

Specimen Ref	Original surface condition	Lubricant No.	Approach velocity, inches/min	Final length, inches	Maximum load, tons
Pr 1	Polished	22	9	0·880	28·5
Pr 2	Polished	Nil	9	0·890	27·8
Pr 3	Fine turned	19	9	0·918	23·0
Pr 5	Fine turned	19	1	0·905	25·0

A criticism which might be levelled at the test procedure is that the top die was placed in contact with the lubricated specimens before the load was applied. During the interval between setting up and loading (approximately 15 seconds) the lubricant layer might be expected to thin excessively. Check tests were carried out to determine the rate at which the film thinned under the weight of the top die alone. The specimen Pr 5 with a maximum surface profile variation from flatness of 0·0002 inch (see Fig 4) was placed on a surface plate, unlubricated, with the top die resting on it. Three vertical side guides were set up touching the die lightly to locate it in position, and the dial gauge was mounted to contact the upper face of the die, giving a datum reading for die-to-specimen contact in the absence of a lubricant. The die was then lifted out of contact and the selected lubricant applied liberally to the top face of the specimen. On replacing the die in contact, the dial gauge reading was recorded against time, giving the curves shown in Fig 6 for the two lubricants used. The curves show the means of two sets of readings with each lubricant which were coincident to within 0·00005 inch at all times. They demonstrate that after moderate intervals of time, films of substantial thickness were still present at the interface.

RESULTS OF TESTS

Table II records the surface appearance of the specimens after testing.

TABLE II

Specimen Ref	Surface appearance after compression
Pr 1	Matt, with a bright annular peripheral contact ring
Pr 2	Highly reflective
Pr 3	Matt, superimposed on original fine machining marks, with a bright annular peripheral contact ring
Pr 5	As for Pr 3

Figs 1 (*a*), 2 (*a*), 3 (*a*), and 4 (*a*) show the specimen surface profiles measured before testing, across the three diameters A, B, and C disposed at 120°. The

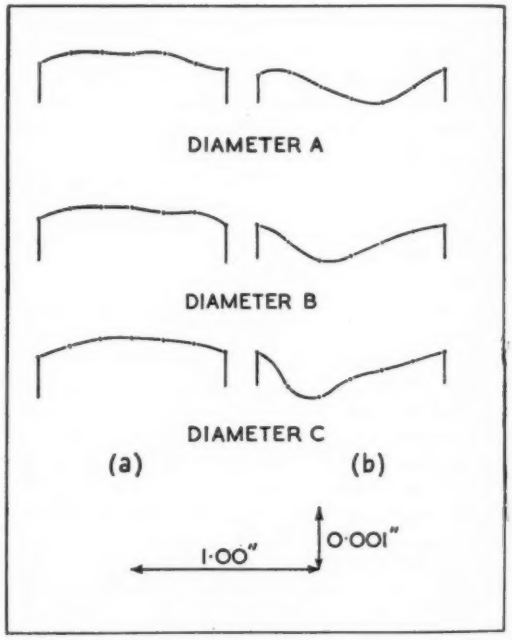

FIG 1
INITIAL AND FINAL PROFILES—SPECIMEN Pr 1

faces are not flat, the variation being up to 0·0005 inch in the worst case (Fig 2). It appears that the edges of the polished specimens, Figs 1 and 2, have been rounded slightly during the polishing operation, despite the fact that a polishing jig was used. The fine turned surfaces, Figs 3 and 4, are more random in

roughness. It should be appreciated, however, that the surfaces are greatly distorted in plotting due to the large vertical to horizontal magnification ratio of 1000 : 3.

The profiles measured after compression are shown in Figs 1 (*b*), 2 (*b*), 3 (*b*), and 4 (*b*). In the presence of both the lubricants, and using the higher approach speed (Figs 1 and 3), the final profiles are concave upwards, the depression in specimen Pr 1 being more pronounced than that in Pr 3. With no lubricant at the higher approach speed (Fig 2), and with the lesser viscosity lubricant at the lower approach speed (Fig 4),

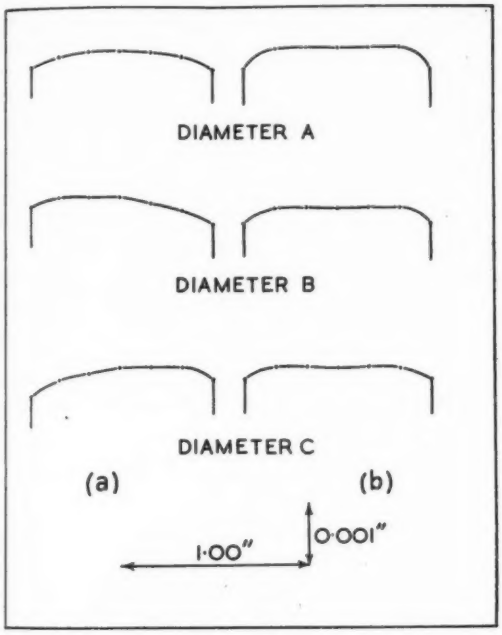

FIG 2
INITIAL AND FINAL PROFILES—SPECIMEN Pr 2

the final profiles are convex upwards, substantially flat, with a slight fall away at the periphery.

Fig 5 shows clearly the depression in specimen Pr 1 as a light gap between the specimen surface and a straight edge laid across it.

DISCUSSION

Before discussing the significance of the final profiles and surface appearances it is necessary to consider what modification to these profiles could result from elastic deformation of the die during loading, and recovery of the specimen on removal of the load. When concerned with profile measurements as small as those currently being considered, it is possible that these effects might mask the hydrodynamic film effect.

It can be shown [5] that the elastic deformation of a

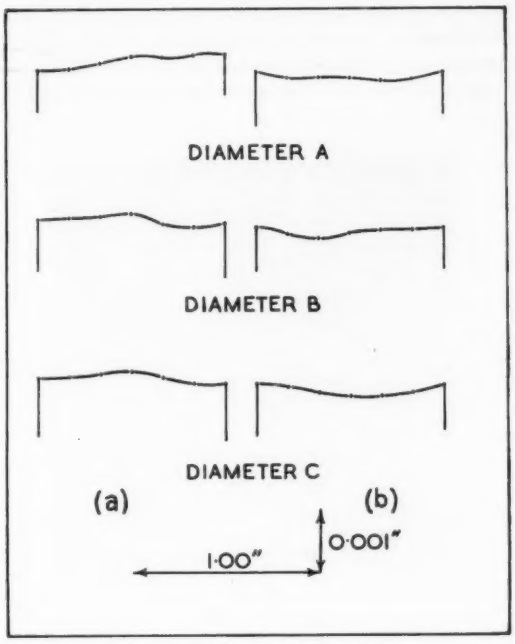

FIG 3
INITIAL AND FINAL PROFILES—SPECIMEN Pr 3

semi-infinite overlapping die during loading is non-uniform. At a load of 25 tons, assuming a uniform distribution of pressure over a 1-inch diameter speci-

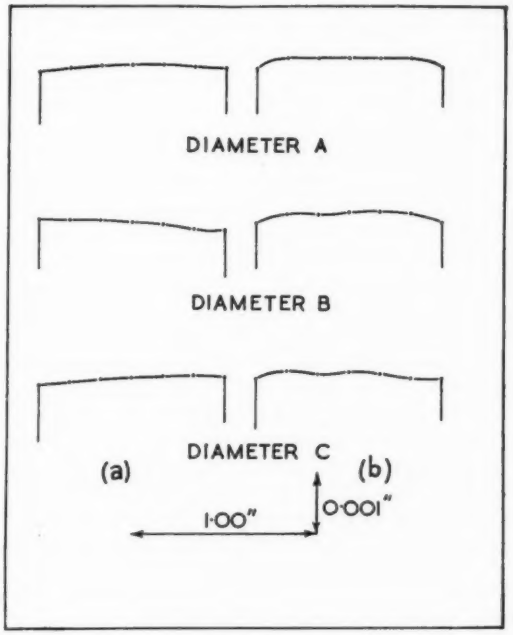

FIG 4
INITIAL AND FINAL PROFILES—SPECIMEN Pr 5

men, the elastic deformation of such a die varies from 0·002 inch at the axis to 0·0013 inch at the specimen periphery. Whilst the die will recover completely on removal of the load, the specimen, having entered the

FIG 5

SURFACE DEPRESSION IN SPECIMEN Pr 1 AFTER COMPRESSION, SHOWN AS A LIGHT GAP (×2)

plastic range, is likely to retain at least a part of the differential strain, recovering to a crowned profile. In addition, on release of the load the specimen itself will experience a general recovery. If this were uniform and elastic, a recovery of 0·0024 inch after loading to 25 tons would occur. However, each element of the material is not subjected to the same complex stress system. Three directional stresses, normal, tangential, and radial, act on each element, and plastic yielding at any position will occur at some combination of the three principal stresses satisfying an accepted yield criterion (*e.g.* von Mises–Hencky [6]). Since the radial stresses reduce to zero at the free edge (periphery) of the specimen, it follows that initial yielding will progress from the periphery towards the axis on increase of load. The recovery of a loaded specimen under these conditions would be expected to be non-uniform, the profile again being crowned. Thus it is unlikely that the final measured profiles are the same as those present under the deforming load, the recovery effects combining to reduce or eliminate any depression originally present.

The final depressed profiles of specimens Pr 1 and Pr 3, together with their surface appearances described in Table II, indicate that in these cases die-to-surface contact was almost entirely absent. During the die approach the hydrodynamic pressure built up in the film was sufficient to initiate plastic deformation before metal contact occurred. The pressure distribution falling from a peak central value to atmospheric at the periphery caused surface dishing before initial metal contact took place at the peripheral boundary. This metallic seal prevented further lubricant expulsion and maintained a continuous hydrostatic film throughout compression, causing surface deterioration to occur, influenced largely by the microscopic metallurgical and mechanical properties of the material as previously described.[3] On release of the load, the recovery of the specimens was insufficient to obscure the depressed contours. The residual maximum depression depth in the specimen Pr 1 was of the order of 0·0007 inch, compared with 0·0002 inch in the case of Pr 3. This is expected, as with the higher viscosity lubricant No. 22, for a given approach speed, a fluid pressure distribution sufficient to initiate plastic deformation would occur at a greater film thickness than with the less viscous (No. 19) lubricant.[4]

The specimens Pr 2 and Pr 5 both show final crowned profiles, but their surface appearances differ substantially. For the former, with no lubricant present, surface matching of the die and specimen occurred, relative motion at the interface leading to burnishing of the specimen, giving its highly reflective appearance. The presence of the crown is attributed entirely to the recovery effects described previously. For the latter it is considered that a shallow depression and lubricant film was present during compression appropriate to the low approach velocity, and that via this film the surface deterioration observed took place. On release of the load, however, the specimen recovery

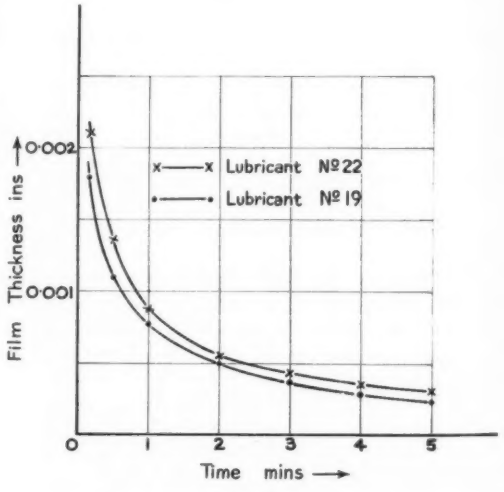

FIG 6

THINNING OF LUBRICANT FILM WITH TIME

completely masked the depression. Further evidence supporting this comes from Figs 2 (*b*) and 4 (*b*), which show the final crown height above the specimen edge of the unlubricated specimen to be greater than in the lubricated case.

CONCLUSIONS

The present work gives experimental support to the previous [4] theoretical hypothesis that continuous lubricant films may be present during bulk metal deformation processes, even when geometrical configurations of the tool and metal surfaces are apparently unfavourable and strain rates are moderate. The presence of such films has two major consequences:

(1) The surface deformation of the metal is not influenced by the die surface, and may be such as to cause a marked deterioration visually.

(2) The total area of the die-to-metal contact is considerably reduced, much of the load being supported through the lubricant film itself. This suggests that the frictional resistance at the interface originates partly from die-to-metal contact and partly from the shear properties of the lubricant itself, under the current conditions of temperature and pressure. In these circumstances it is reasonable to assume that measured values of coefficient of friction will be smaller during deformation processes than during the conventional sliding tests, an experimental observation previously made by Whitton and Ford.[2] It follows too that there is no reason to suppose that during a deformation process the coefficient of friction will necessarily remain constant. As deformation proceeds, thinning of the trapped, pressurized film, combined with the surface roughening effect, will bring into contact more metallic areas, changing continuously the ratio of the contributions made by the die-to-metal contact areas and lubricant film shear.

ACKNOWLEDGMENT

The author acknowledges the assistance given by C. A. Hopkins, senior technician in the Engineering Department, University College, Cardiff, during the experimental work reported in this paper and the companion paper published in the *Journal* in March 1960.

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THE DETERMINATION OF WATER IN GASEOUS HYDROCARBONS: FISCHER REAGENT METHOD *

By THE PETROLEUM GASES PANEL OF THE INSTITUTE OF PETROLEUM †

INTRODUCTION

From time to time during recent years, the Petroleum Gases Panel has given consideration to the various methods that are available for the determination of moisture in hydrocarbon gases. A knowledge of the water content of a gas stream may sometimes be required for routine control purposes and is often of value during pilot plant investigations into specific refinery processes. From the point of view of LPG production, the presence of moisture in the marketed product can be troublesome to the consumer, since under low temperature conditions hydrate formation can lead to blockages in valves and regulators. In the U.K., climatic conditions are such that this is fortunately not a serious problem and in consequence most current LPG specifications in Britain merely require that the marketed gas be free from "mechanically entrained water." This clause is not unduly restrictive on the refiner, but since C_3 and C_4 hydrocarbons, as produced by distillation from crude oil, normally contain undesirable sulphur compounds, such as mercaptans, which must be reduced in concentration by refining processes (*e.g.* caustic wash) involving the use of aqueous solutions, LPG as normally marketed may well have a water content equivalent to the saturation value of water in the liquid hydrocarbons concerned. The production of LPG with water contents lower than this value is expensive and necessitates the installation of elaborate gas drying equipment.

It is also important to bear in mind that when a gas is drawn, as is normal practice, from the vapour space of a vessel-containing LPG under pressure, the relative concentration of the water in the gas and liquid phases changes and so an analysis of the moisture content of such gas will give an entirely different result from that which would be obtained if a sample was drawn direct from the liquid phase. The moisture content in one phase may be converted to that in the other phase by using the distribution coefficient appropriate to the sampling temperature.[1,2]

In view of the above considerations there has, as yet, been no clear-cut demand for publication of an IP standard method for water in gases. On the other hand, as already mentioned, refinery and research laboratories often find it necessary, for purposes of internal interest, to examine gaseous samples for water content. It has been found that in such circumstances the method that is most commonly employed is based on the use of the well-established Karl Fischer reagent, and although individual details of procedure may vary from laboratory to laboratory, as for example in the design of reaction vessel, gas flow rate, etc., a considerable background of experience on this type of analysis has now been gained and shared by the members of the Petroleum Gases Panel. In view of this, a representative procedure for determining moisture content in gases by means of the Fischer reagent has been agreed upon by the Panel. The method admittedly has its limitations; it is subject, for example, to interference by various sulphur compounds, and difficulties can arise if it is applied to samples of C_3 hydrocarbons. However, the Panel felt that, provided such limitations are borne in mind, the method might well be of interest to other laboratories, and in particular to those with little or no previous experience of this type of analysis. The agreed procedure is therefore published below for general information. It should only be regarded as tentative, and comments and suggestions would be welcomed by the Panel. These should be sent in the first instance to the Technical Assistant, Institute of Petroleum.

METHOD

Scope

1. This method is suitable for the determination of water in hydrocarbon gases and some LPG (liquefied petroleum gases) subject to the provisions mentioned later. The method is applicable to concentrations of water from a few parts per million up to 2 mg 100 ml gas. It is not suitable for the determination of

* MS received 13 May 1960.

† The present membership of the Petroleum Gases Panel is as follows:

J. H. D. Hooper (Chairman), The British Petroleum Co. Ltd
K. R. Garrett (Deputy Chairman), Calor Gas (Distributing) Co. Ltd
C. A. Miller (Secretary), The British Petroleum Co. Ltd
C. Akehurst, Petrochemicals Ltd
C. E. Cobb, Esso Petroleum Co. Ltd
A. G. Inglis, BP Refinery (Grangemouth) Ltd
T. E. Jenkins, BP Refinery (Llandarcy) Ltd
W. K. Leadill, Imperial Chemical Industries Ltd
S. G. Perry, Esso Research Ltd
A. F. Pyrah, Mobil Oil Co. Ltd
D. Rawlinson, Shell Refining Co. Ltd
J. C. Street, Shell-Mex and B.P. Ltd

[1] *Oil Gas J.*, 16.9.48, **47** (20), 133.

[2] Poettmann, F. H., and Dean, M.R. *Petrol. Refin.*, 1946, **25**, 635.

water in liquid mixtures which are substantially C_3 hydrocarbons unless the sampling vessel line is suitably heated. Substances, other than water, which react with Karl Fischer reagent should be absent, although concentrations of either hydrogen sulphide or mercaptans below 10 grains S/100 cu ft (0·023 mg S/100 ml gas) can be tolerated if the quantity present does not exceed the water content. In this case a correction must be applied for the known hydrogen sulphide or mercaptan content.

Outline of Method

2. The gas sample is passed through a small titration vessel containing ethylene glycol and a known excess of Karl Fischer reagent. The water content is calculated from the volume of gas required to consume (neutralize) the Fischer reagent.

Apparatus

3. The apparatus assembly is shown in Fig 1.

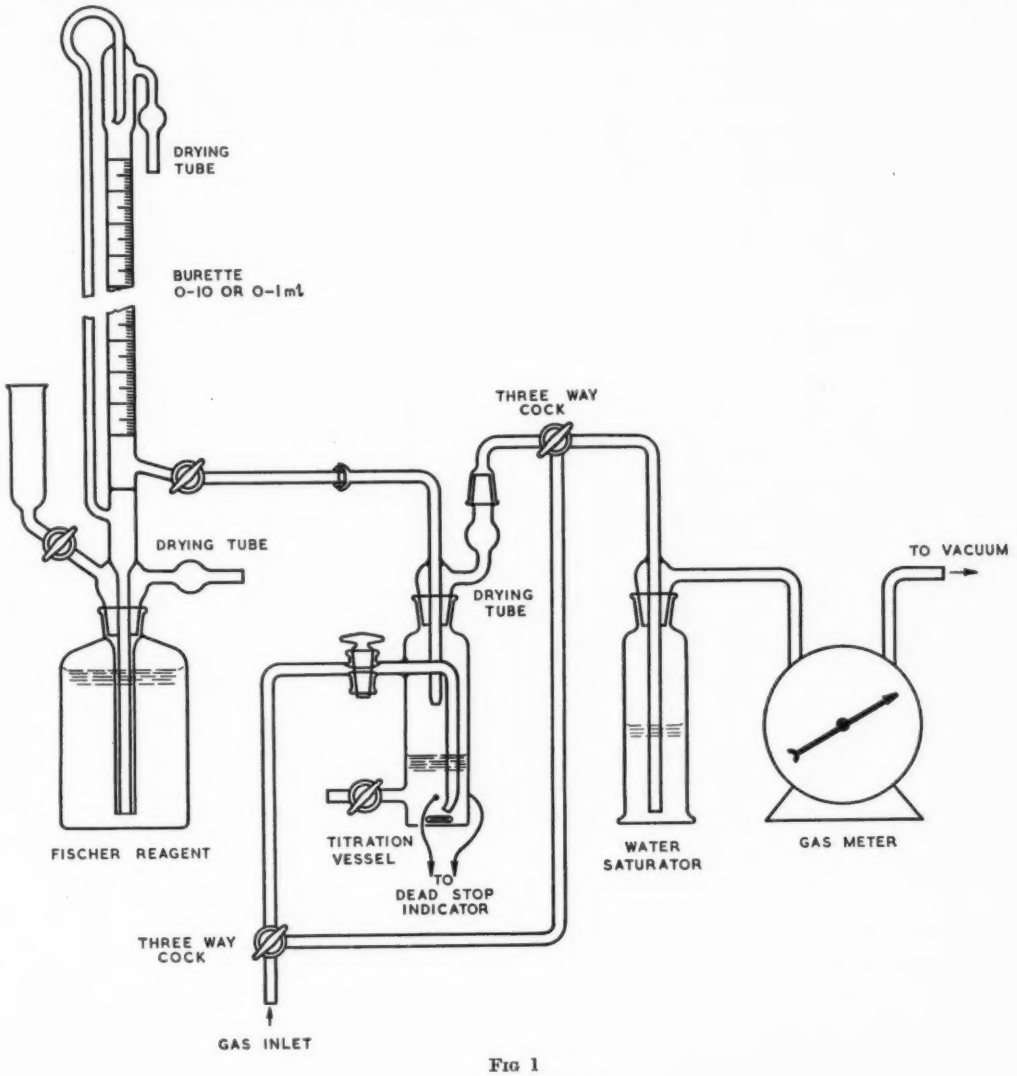

FIG 1
APPARATUS ASSEMBLY

(*a*) *Dead-stop Indicator Apparatus.* A description of a circuit is given in Fig 2. This apparatus has a potential of approximately 0·8 volt across the two platinum electrodes and a sensitivity of approximately 50 millivolts. Any dead-stop apparatus with comparable sensitivity will be suitable.

(*b*) *Titration Vessel*—a glass vessel fitted with a pair of platinum electrodes, a gas inlet and outlet as

described in (*c*), and a drain. The dimensions of the vessel are not critical, although the enclosed volume should not exceed 100 ml. A convenient type of vessel is shown in Fig 3 (*a*) and an alternative design in Fig 3 (*b*).

(*c*) Burette assembly consisting of three automatic burettes, two of 10 ml capacity (calibrated in 0·05 ml) and one of 1 ml capacity (calibrated in 0·01 ml). The

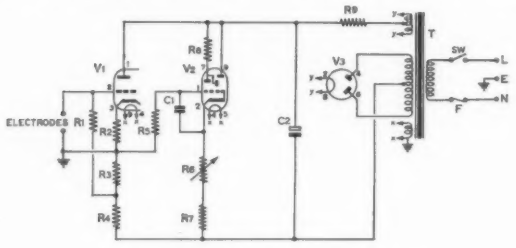

FIG 2

CIRCUIT OF ELECTROMETRIC TITRIMETER

R1—100 kΩ	C1—0·1 μF
R2— 1 kΩ	C2—16 μF
R3— 1 kΩ	V1—ECC 83
R4— 40 kΩ	V2—EM 81
R5—470 kΩ	V3—5Y3 or nearest equivalent
R6— 50 kΩ	T—Transformer
R7— 10 kΩ	240–0–240 v, 40 mA
R8—470 kΩ	6·3 v 2 A, 5 v 2 A
R9— 15 kΩ	F—Fuse, 1 amp

R6 Variable wirewound 5 w
R9 Wirewound ±5% 5 w
All other resistors ±20% 1 w

Note: This circuit has been devised by the Hydrocarbon Analysis Panel of the IP Analysis Sub-Committee. It will be published in the 1961 Edition of IP Standard Methods in place of the titrimeter circuit currently shown as Fig 1 of IP 130/56.

burette tips should be detachable (ball and socket joints) and provided with ground glass stoppers to fit the titration vessel. The stoppers should be of the hollow type containing a gas outlet equipped with a drying tube.

(*d*) Magnetic stirrer including glass covered paddle.

(*e*) *Gas Meter*—wet type. A 0·5 litre/rev is recommended.

(*f*) A suitable sample bomb for sampling LPG in gaseous or liquid form.

Materials

4. (*a*) *Ethylene Glycol*—anhydrous. The water content should not exceed 0·1 mg/100 ml ethylene glycol (see Note 1).

(*b*) *Pyridine*—anhydrous (see Note 1).

(*c*) *Methyl Cellosolve*—anhydrous (see Note 1).

(*d*) *Karl Fischer Reagent*—strong. 1 ml equivalent to 5 mg of water. For one litre of solution, dissolve 133 g of c.p. iodine in 425 ml. of c.p. pyridine in a dry glass-stoppered bottle, add 425 ml of anhydrous methyl cellosolve (monomethyl ether of ethylene glycol), and cool in an ice-water bath. Add in small increments and with constant swirling 70 ml of anhydrous liquid sulphur dioxide from a graduated cylinder (or 102 g of gaseous sulphur dioxide from a

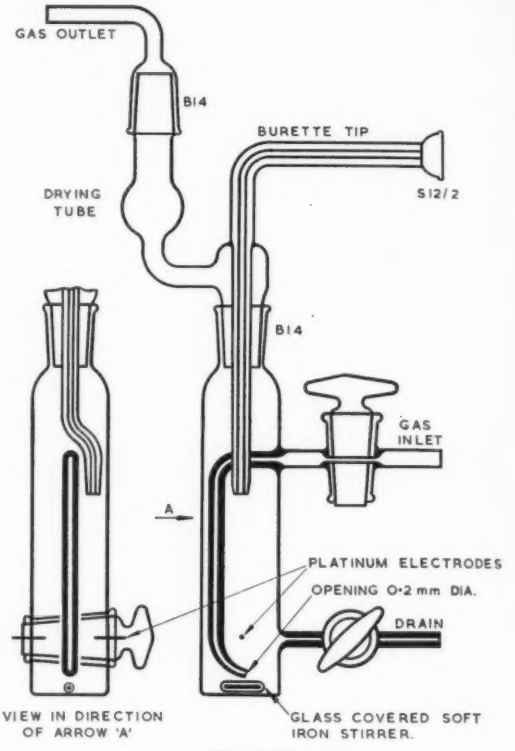

FIG 3 (*a*)

TITRATION VESSEL

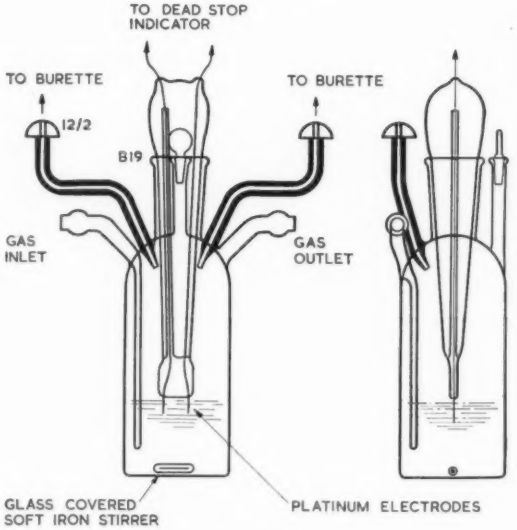

FIG 3 (*b*)

ALTERNATE TITRATION VESSEL

gas cylinder). Mix thoroughly and dispense from burette assembly. The water equivalent (F) of this reagent is approximately 6 mg of water per ml of reagent.

Standardize the reagent in the following manner: Introduce 10 ml of 50/50 ethylene glycol–pyridine mixture into the titration vessel. Stopper the titration vessel, start the stirrer, and add Fischer reagent until the end point is reached. Add from a Lunge weighing pipette approximately 0·1 g of water and, with constant stirring, titrate with Fischer reagent until the end point is reached again. Calculate the equivalency factor F as mg of water per ml of the Fischer reagent.

Note: Alternatively sodium acetate (trihydrate) may also be used to standardize the reagent.

(*e*) *Dilute Karl Fischer Reagent.* Dilute the strong reagent with methyl cellosolve until the water equivalent is approximately 1 mg of water per ml of reagent. Determine the exact water equivalent of this solution in a similar manner to the standardization of the strong solution.

(*f*) *Desiccant*—Drierite (anhydrous calcium sulphate). (6–10 mesh is recommended.)

NOTE 1: If the water content of the methyl cellosolve, ethylene glycol, or pyridine exceeds 0·1 mg/100 ml liquid, dry these solvents by fractionation. The removal of approximately 5 per cent in a six-plate column is suitable. Use the remaining 95 per cent.

Preparation of Sample

5. (*a*) Great care is necessary to ensure that no water is taken up from plant sample lines or valves. All connexions should be as short as possible and, in the laboratory, glass tubing should be used wherever possible with connecting joints of short lengths of polythene. Rubber and PVC tubing must not be used.

Connexions from plant sources should be made by means of copper lines thoroughly heat-dried before use. Before sampling, all connexions must be flushed very thoroughly, with the gas under test. If this is not done, high results will probably be obtained on the first gas samples tested.

(*b*) LPG may be sampled in vapour phase or in liquid phase. As the two procedures will give quite different results, depending on the partition of water between the two phases, the method of sampling must be reported with the results obtained.

Due to the formation of hydrates at the sample bomb exit valve, estimation of water in liquid C_3 hydrocarbons by this method gives low results unless hydrate formation is prevented by heating the exit valve.

Procedure

6. (*a*) Fill one of the 10-ml burettes with strong Fischer reagent, and the other 10-ml and the 1-ml burette with diluted Fischer reagent. Pack the drying tubes with Drierite (anhydrous calcium sulphate). The burette to be used depends on the water content of the sample. The following table may serve as a useful guide.

Water content (mg/100 ml gas)	Burette	Reagent
0·2–2·0	10 ml	Strong
0·03–0·3	10 ml	Dilute
Less than 0·03	1 ml	Dilute

(*b*) Place 5–10 ml of ethylene glycol in the titration vessel, start the stirrer, and titrate with Fischer reagent to the dead stop end point.

(*c*) With the three-way stopcocks in the " by-pass " position, flush the gas inlet lines with the gas, allowing sufficient gas to pass through to ensure that all residual water in the lines has been removed.

NOTE 2: The " by-pass " position of the three-way stopcocks is the position when the gas lines to and from the titration vessel are blocked off.

(*d*) After the flushing procedure prepare the cell by adding a small excess of Fischer reagent. Re-adjust the positions of the three-way stopcocks and allow the gas to pass through the solution in the titration vessel until the end point is reached as shown by the " magic eye " indicator.

(*e*) Add an appropriate quantity of reagent (see Note 3) and pass the gas through at a rate of approximately 10 litres/hour for at least 20 minutes. When the end point is reached, stop the flow of gas and read the volume of gas as indicated by the meter.

NOTE 3: If the time taken for the test is less than 20 minutes under the stated conditions the test must be repeated using a larger quantity of Fischer reagent. A rate of 10 litres/hour is specified to avoid mechanical loss of reagent, but the rate of absorbing water is important. Conditions are obviously different passing an almost dry gas and a saturated gas at 10 litres/hour. The " 20 minutes " sets some control on the " water rate."

Calculation

7. (*a*) Calculate the volume of the sample in litres of dry gas at NTP by means of the following equation

$$\text{Gas Volume } V \text{ (litres)} = Vo \times \frac{273}{(273 + t)} \times \frac{B + (P - p)}{760}$$

where Vo = volume of gas as read from the gas meter, litres;
t = average temperature of the gas at the gas meter, °C;
B = barometric pressure, mm Hg;
P = excess pressure of the gas in the gas meter, mm Hg;
p = vapour pressure of water at t° C, mm Hg.

(*b*) Calculate the water content of the gas (hydrogen sulphide and mercaptan-free) as mg/100ml gas on a dry gas basis by means of the following equation:

$$\text{Water (mg/100 ml gas)} = \frac{T \times F}{10V}$$

where T = volume of Fischer reagent used in the determination, ml;
F = water equivalent of the Fischer reagent, milligrams of water per ml of reagent; and
V = volume of sample in litres of dry gas at NTP (see 7 (*a*)).

(*c*) If the gas contains hydrogen sulphide and/or mercaptans in concentrations below 10 grains S/100 cu ft (*ca* 200 mg/m³) and in quantities not exceeding the water content, calculate the latter by means of the following equation:

(i) Water (mg/100 ml gas) =

$$\frac{TF}{10V} - \frac{12{\cdot}84a + 6{\cdot}42b}{10{,}000}$$

where a = content of hydrogen sulphide sulphur in the gas, grains/100 cu ft;
b = content of mercaptan sulphur in the gas, grains/100 cu ft.

or

(ii) Water (mg/100 ml gas) =

$$\frac{TF}{10V} - \frac{0{\cdot}56x + 0{\cdot}28}{10{,}000}$$

where x = content of hydrogen sulphide sulphur in the gas, mg/m³;
y = content of mercaptan sulphur in the gas, mg/m³.

Precision

8. Duplicate results by the same operator should not differ by more than the following amounts:

Water content (mg/100 ml gas)	Repeatability
2	1% of mean
0·03–2	5% of mean
0–0·03	10% of mean

OBITUARY

PIERRE MARIE EDMOND SCHMITZ

1881–1959

It is with regret that we record the death in December last of Dr P. M. E. Schmitz, a member of the Institute since 1921 and one who first joined the oil industry as long ago as 1906.

Pierre Schmitz was born in Strasbourg on 1 August 1881, was educated at the University of Nancy, and received his diploma in chemistry at the Ecole de Chimie de Mulhouse in 1903. In 1907 he took his degree of doctor of science at the University of Grenoble.

In 1906 he became joint director in the petroleum research laboratory of the Societi Steaua Romana, and in 1907 was appointed chief chemist of the Nobel Society in Baku. In 1914 he was appointed director of the Dürrenbach refinery at Pechelbronn in Alsace and in 1918 went to Ploesti, Rumania, to take charge of the Vega refinery. Three years later he returned to Rumania with the Concordia Company, and in 1923 joined the staff of Petrofina in Belgium. He retired in 1949, and at that time was one of the oldest members to leave the service of the Petrofina Company.